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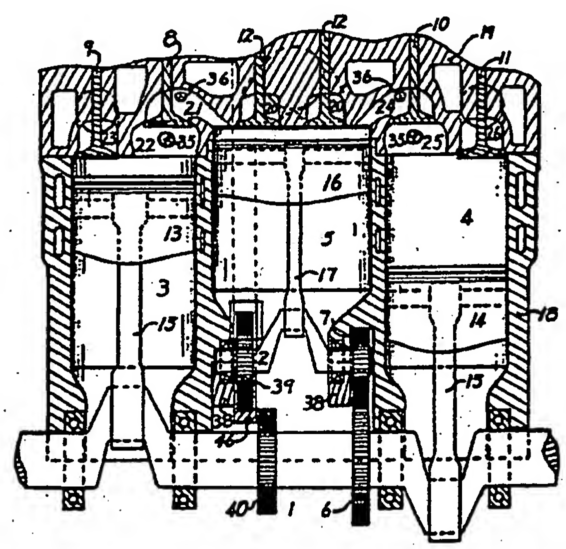

(57) Abstract

An internal combustion engine power unit comprises two power cylinders (3, 4) spaced equidistant about a pumping cylinder (5). All cylinders operate on two-stroke cycles, the power cylinders (3, 4) having a phase difference of 180°. Power piston assemblies (13, 14) in the power cylinders (3, 4) drive crankshaft (1). Pumping piston (16) and separate crankshaft (2) are driven at twice the cyclic speed of the power pistons (13, 14) and crankshaft (1) through gear train (6, 7) between the respective crankshafts (1, 2). Air inducted into pumping cylinder (5) via intake ports (20) is compressed and passed alternately to power cylinders (3, 4) via valve controlled transfer passages (21, 24). All valves, ports and gas passages are found in a cylinder head (19). Timed fuel injection and ignition are provided. An engine may comprise one or more power units. There is also disclosed a turbo-charged diesel engine comprising two power units in "V" configuration.

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RECIPROCATING PISTON ENGINE WITH PUMPING AND POWER CYLINDERS

TECHNICAL FIELD

This invention relates to reciprocating piston internal combustion engines of the type wherein, pumping and power cylinders are operated on two stroke cycles.

BACKGROUND ART

Engines of this type have been disclosed in numerous prior art which have intended to improve the efficiency and or power to weight ratio thereof. US,A, PATENT NO 1,881,582 shows a design which has a pumping cylinder driven at twice the cyclic speed of and alternately suppling a intake scavenging charge to two power cylinders, via transfer ports which communicate with the lower cylinder walls of the power cylinders, hence being timed by the power pistons. Although this design marginally increases the scavenging efficiency attainable and as compared to crankcase compression type two stroke engines, this design has and retains numerous efficiency problems of, including the fundamental inefficiency of, the conventional two stroke engine. The said inefficiency results from the opening of the transfer ports in the lower cylinder walls and which reduces the volume through which expansion occurs with the said reduction being used instead for a half of the transfer scavenging phase. Furthermore this design, due to the said transfer to the lower cylinder walls, has no potential for significant efficiency gains to be attained if valve controlled constant volume combustion chambers are to be used.

A second type of engine which has pumping and power cylinders operating on two stroke cycles and which have intended to overcome the above said undesireable features are typically disclosed in U S PAT NO`S : 3,880,126 and 4,458,635. These designs have the pumping cylinder transferring the intake charge through valve timed ports which open into the power cylinder head section. U S PAT NO : 3,880,126, utilizes a combustion chamber which is in constant communication with the power cylinder and which has an excessive number of components whilst overall efficiency and power output are severley limited by a poor scavenging efficiency which primarily results from

the long transfer scavenging phase required of the design. This further exacerbats the obvious power to weight ratio limitations of the design. U S PAT NO: 4,458,635, utilizes a valve controlled constant volume combustion chamber which foregoing the supercharging system used that results in a similar said fundamental inefficiency, increases the scavenging and combustion efficiency and hence overall efficiency is also maginally increased. Subsequently, only an average power to weight ratio results whilst an excessive number of components is still a major problem.

A further design of engine which shares similiar cylinder, port and valve locations of the presented invention but which is outside of the technical field of this invention in that the power cylinders operate on four stroke cycles, is typically shown in GB,A PATENT NO 2071210. As such the pumping cylinder is used only as a supercharging device and is not necessary for the operation of the engine as is required in the presented invention.

DISCLOSURE OF INVENTION

The presented invention discloses a novel design of engine which, significantly increases the thermal efficiency and power to weight ratio of all above said types of engines and increases the scavenging efficiency of the above said engine types which are within the field of this invention, and decreases the number of components required for, the above said second type of engine.

The principal object of this invention therefore describes a engine which has one or more units with a unit having, a pumping cylinder with a pumping piston reciprocable therein and two power cylinders which have power pistons reciprocable therein, whilst all said cylinders operate on two stroke cycles and with the pumping piston being driven by means to reciprocate at and cyclicly operate at twice that of the power pistons of the power cylinders and with the intake charge of consecutive pumping cylinder cycles, being transferred alternately to the said power cylinders through transfer ports which communicate the head section of the pumping cylinder with the head section of each of the power cylinders. Each said

transfer port has communication between the said cylinders timed by atleast transfer valve means which, time communication between each transfer port and the power cylinder which it opens into. The pumping cylinder induces, thereinto and through a intake valve timed intake port which passes through the said head thereof, atleast a major portion of the intake charge on its increasing volume stroke before on its decreasing volume stroke, the said charge of consecutive cycles thereof is transferred alternately to each of the power cylinders and or their respective combustion chambers through the said transfer valve timed transfer ports. The intake charge is atleast 60% of the air used in combustion whilst the head section may include the upper portion of the cylinder walls wherein, less than one third of the cylinder volume is provided. A power mainshaft causes reciprocation of the power pistons which relative to each other, are phased or phased about, one stroke apart, whilst a pumping mainshaft causes reciprocation of the pumping piston at the above said rate. Valve timed exhaust ports also exit the power cylinders through the said head section and provide for the expanded gases thereof, to be exhausted therefrom. The said combustion chambers may be in constant communication with its respective power cylinder or communication therebetween may be controlled by a secondary valve which is timed to provide for constant volume combustion for atleast a portion of the time required for said combustion whilst in any case, from hereinafter and above, the opening of a respective transfer valve of a particular power cylinder is refered to as opening into that said power cylinder.

Preferable, the pumping piston of a said unit, is equally distanced to the power cylinders thereof and leads the power piston of the power cylinder which the intake charge is to be or is being, transferred into, to the 'top dead centre' (from hereinafter refered to as 'TDC') position by, less than 60 % of the time the power cylinder piston is moving towards the said position whilst the pump mainshaft is driven by means, from the power mainshaft or the output shaft of the engine. It is further preferred that the respecitve transfer valve is closed before combustion initiates in that power cylinder whilst preferred valve timings which allow for the efficient operation

of the said engine are also stated.

A further object of this invention has the engine just described being optionally modified by various improvements thereto and which have; the pumping cylinder mainshaft being located axially above the power cylinder mainshaft and or, to have the valve train actuating means and or other auxilary device, being driven from means provided on or being located on the pumping cylinder mainshaft or, on the power cylinder mainshaft between the said power cylinders and which provides for a compact engine and unit to be achieved; a variable valve timing mechamism which varies atleast the closing time of the exhaust valve so that its closing time may be varied to allow efficient operation under transient operating conditions. Desirable combustion chamber designs of both abovesaid combustion chamber types with the transfer and secondary valves being poppet type valves and with desirable locations and timings thereof are further objects of this invention. A still further object of this invention has enviable V configurations and turbocharged designs of the novel engine whilst a further object has the pumping cylinder utilizing crankcase compression thereof to improve the charging efficiency thereof.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG 1- is a top schematic view of the preferred design which is a inline single unit and showing the cylinders, ports, combustion chamber and valve opening locations thereof.

FIG 2- is a cross sectional view taken along line A-A of FIG 1 but around the piston crankshaft mechanism and with the lower crankcase removed.

FIG 3- is a valve timing diagram of the preferred design in power cylinder crank angle degrees with the lines indicating valve open times and with the TDC position shown thereon being the TDC position of the first power piston.

FIG 4- Shows an alternative design which has two units, being in a V configuration and utilizing turbocharging and crankcase compression of the pumping cylinder. One unit or bank of cylinders is shown as a end view with the other unit shown as a cross sectional view taken along line B-B of FIG 5 but around the piston crankshaft mechanism thereof and with partial

hidden detail shown and the lower power crankcase and lower RH side pumping cylinder crankcase being removed.

FIG 5- is a top schematic view of the sectioned unit of FIG 4 and shows the cylinders, ports, combustion chambers and valve opening locations thereof.

FIG 6- is a valve timing diagram of the alternative design shown in FIGS 4 and 5 and uses the same features as described for FIG 3.

FIG 7- is a end shematic view of an alternative V configuration and which shows the cylinder and crankshaft locations thereof.

MODES FOR CARYING OUT THE INVENTION

Refering to all modes for carrying out the present invention, each said unit has a pumping cylinder 5 with a pumping piston 16 reciprocable therein and first and second power cylinders, respectively 3 and 4 with first and second power pistons respectively 13 and 14 reciprocable within their respective power cylinders. All cylinders of a unit share a parallel axis and a common block 18 and a common head 19, whilst the pumping cylinder is evenly distanced to each of the power cylinders. A pumping crankshaft 2 and pumping conrod 17 cause reciprocation of the pumping piston 16 and a power crankshaft 1 and power conrods 15 cause reciprocation of the said power pistons. Each said crankshaft is supported for rotation by bearing means whilst journal means which are not shown in the drawings, provide pivotal movement at the conrod crankshaft pivots and the conrod piston pivot. A pump drive gear 7 which is fixed to each of the pumping crankshafts 2, cooperates with, is driven by, and is one half the diameter of, the power crankshaft gear 6 which is fixed to the power crankshaft 1. This gear arrangement then provides for the pumping pistons 16 to be reciprocated at and cyclicly operated at, twice that of the power pistons. The phasing of the power pistons of a unit relative to each other, is one hundred and eighty power 'crankshaft crankangle' (from hereinafter is refered to as 'CA') degrees. The first and second transfer ports respectively 21 and 24, remain in constant communication with the pumping cylinder. The crankshafts for carrying out all

modes of the invention, are of the one piece type whilst all conrods are of the two piece type and bolt on to the respective crankshafts from the undersides thereof, for pivotal movement therearound. Of course the components and auxiliaries not illustrated and refered to, and which are required for the efficient operation of the engine, are included in all modes for carrying out the invention whilst water cooling passages are shown in the sectioned walls of FIGS 2 and 4 but are not numbered to reduce cluttering thereof. Furthermore, the respective components of the first and second power cylinders are respectively refered to as the first and second said components, or they are refered to as the respective components of the power cylinder of which the description is directed to.

Refering now to FIGS 1-3, the preferred design or mode for carrying out the invention, is a naturally aspirated inline version and with the pumping cylinder 5 being located in the middle of the first and second power cylinders, respectively 3 and 4. The pumping crankshaft 2 is accessed and held in place by pumping crankshaft caps 38 which bolt into the engine block 18 whilst the power crankshaft 1 is accessed and held in place by the lower crankcase which is removed in the FIG 2. The phasing of the pumping piston 16 relative to the power pistons 13 and 14 has the pumping piston leading the piston of the power cylinder which the intake charge of that particular pumping cylinder cycle will be transfered into, to TDC, by forty power CA degrees.

The preferred design has all intake, transfer, and exhaust valves being poppet type valves. The first and second combustion chambers respectively 22 and 25, remain in constant communication with their respective power cylinder and each has a spark plug 35 mounted thereinto and which causes ignition of the combustible mixture therein. Petrol fuel injection means 36 are mounted into each transfer port and inject a predetermined quantity of fuel thereinto as the said intake charge is being transferred into the power cylinder thereof. A first transfer valve 8 times communication between the first transfer port 21 and the first power cylinder 3 whilst a second transfer valve 10 times communication between the second transfer port 24 and

the second power cylinder 4. Two intake valves 12 time communication between the intake port 20 and the pumping cylinder 5. A first exhaust valve 9 times communication between the first power cylinder 3 and the first exhaust port 23 whilst a second exhaust valve 11 times communication between the second power cylinder 4 and the second exhaust port 26. The said exhaust ports lead to an exhaust manifold and eventually to an exhaust pipe whilst the said intake port leads to an intake manifold with air metering means therein provided. All of the said valves are actuated by a single overhead camshaft which has a axis parallel to that of the crankshafts and is positioned directly above all the said valves so as to directly actuate them. The said camshaft is not shown on FIG 2 to reduce cluttering thereof and of the major features therof. The said camshaft is driven by chain means 46 from the camshaft drive sprocket 39 which is fixed to the pumping crankshaft 2. The sprocket on the said camshaft which cooperates with the said chain is a half of the diameter as the said camshaft drive gear, providing for the said camshaft to operate at the same cyclic speed as the power cylinders and as such, single camlobes actuate the transfer and exhaust valves, whilst two camlobes are evenly spaced around the said camshaft where the intake valves are actuated from, so that the intake valves open twice as often as the other valves and which follows the increased cyclic speed of the pumping cylinder. Variable exhaust valve closing event is obtained by a turning block type of variable valve timing mechanism which is not shown for reasons of undue complexity and which allows for the said valves to close between fifty and sevsnty power CA degrees before TDC and depending on engine load and speed. This said variable closing is shown on FIG 3 by the dashed line thereon. The engine oil pump supplies the oil to the engine and is driven from the oil pump drive gear 40 which is fixed to the power crankshaft 1 between the power cylinders.

The method of operation including the valve timings of the preferred design is now described. The intake valves 12 open when the pumping piston moves through to sixty pumping CA degrees after TDC. This allows the compressed intake gas of the previous cycle to expand substantially to atmospheric before

the said valves 12 are opened. With the intake valves 12 opened and the pumping piston moving towards its 'bottom dead centre' (which from hereinafter is refered to as 'BDC') position, the intake air is induced into the pumping cylinder5. As the said piston 16 moves through forty said CA degrees after BDC the intake valves 12 are closed and the induction of the intake air ceases. At the same time the intake valves 12 close, one of the transfer valves 21 or 24 begins to open, initiating the transfer phase to the respective power cylinder of which the said open transfer valve opens into. The said transfer valve then remains open untill the pumping piston 16 moves through to ten said CA degrees after TDC which is shown in FIG 3 and being thirty five power CA degrees before the piston of the said respective power cylinder reaches TDC. The piston of the pumping cylinder then continues towards BDC, and begins a new cycle thereof as is described above and when the intake valves 12 begin to open again at sixty pumping CA degrees after TDC. The intake air of the next said cycle is transfered to the other power cylinder and the intake air of the following said cycle and which is after the said next cycle is transfered to the said respective power cylinder starting a new cycle thereof.

During the first part of the transfer phase to the said respective power cylinder, the exhaust valve thereof is open, providing for the later part of the exhaust phase thereof to occur which has the scavenging of the remaining exhaust gases from the said respective cylinder by the transfering intake air. The exhaust valve of the said respective power cylinder remains open untill the piston thereof moves to between fifty and seventy power CA degrees before TDC. At high load and or high speed, the fuel is injected into the transfer port of the said respective power cylinder during the transfer phase and at low load and or speed, it is mostly injected after the exhaust valve of that power cylinder has closed. With the fuel injected, a spark at the respective spark plug 35 causes combustion to occur about the TDC position. The piston of the said respective power cylinder then moves towards BDC, substantially expanding the gases therein to atmospheric before the exhaust valve begins to open when the said piston is at

forty five power CA degrees before BDC. This then initiates the first part of the exhaust phase being blowdown, and then positive scavenging occurs whilst the piston thereof moves towards TDC untill the transfer valve of that cylinder opens, beginning another cycle thereof and as is described above. The operation of the other power cylinder is the same as that described above for the said respective power cylinder but as is obvious, it occurs one hundred and eighty power CA degrees before and after it occurs in the said respective cylinder.

Refering now to FIGS 4-6, the alternative design or mode for carrying out the invention has two units which are set in a V configuration and with each said unit being one bank of cylinders of the said V. The power cylinders of each unit, are positioned close together with the pumping cylinder 5 of each unit being positioned on the outside of the said V but being central to the power cylinders of its said unit. Constant volume combustion chambers which have communication to their respective power cylinders being timed by secondary valves are used in the alternative design with the first said secondary valve being 27 and the second said secondary valve being 28. A turbocharger 41 is positioned in the middle of the said V with the exhaust manifolds 23 of all power cylinders communicating thereto whilst the exhaust, ports 23 and manifolds 23 share the same number. The pressurised intake manifold 42 leading from the turbocharger 41 communicates with the intake ports of both pumping cylinders whilst the crankcase intake ports 33 of both pumping cylinders is naturally aspirated. A single power crankshaft 1 causes reciprocation of all power pistons whilst each pumping cylinder 5 has its own pumping crankshaft 2. A single power crankshaft gear 6 which is fixed to the power crankshaft, cooperates with the pumping cylinder drive gears 7, fixed to each of the pumping crankshafts. The phasing of the pumping pistons relative to the power cylinders of a respective unit, has the pumping piston leading the said power pistons to TDC by fifty power CA degrees. The phasing of the power pistons of the unsectioned unit relative to the said pistons of the sectioned unit, has the first power piston 3 of the sectioned unit, leading the said first power piston of the unsectioned

unit, by ninety power CA degrees. The power crankshaft 1 is accessed and held in place by the lower power crankcase which is removed in the drawings whilst each pumping crankshaft 2 is accessed and held in place by a pumping lower crankcase 47 which is shown on the unsectioned unit of FIG 4.

The alternative design has all intake, transfer, exhaust and secondary valves being poppet type valves whilst the crankcase intake valves 32 are reed type valves. The first combustion chamber 22 and the first power cylinder 3 has communication therebetween controlled by a first secondary valve 27 whilst the second combustion chamber 25 and the second power cylinder 4 have communication therebetween comtrolled by a second secondary valve 28. Diesel fuel injection means 37 are mounted into each said combustion chamber whilst ignition therin is caused by the temperature and pressure of the combustible mixture therein. Protrusions 31, on the top of each power piston, extend upwards so that they substantially atleast, take up the volumes of each secondary port 29 and 30 which result in an efficiency increase of the engine. The alternative design has each unit having the same intake, transfer, and exhaust valve and port arrangements and functions, as are described for the preferred design although the positioning of some valves and ports is altered. Each said unit has two overhead camshafts which are not shown in the drawings and which are driven by gear means from the pumping cylinder drive gear 7. One of two idler gears 43 cooperates with the said gear 7 whilst the another idler gear 44 cooperates with the idler gear 43 and with the camshaft gear 45 which is the same diameter as the power crankshaft gear 6. The said camshaft gear 45 is fixed to the power camshaft which has single camlobes actuating each transfer, secondary, and exhaust valves whilst another gear which is fixed to the said power camshaft cooperates with a gear which is a half the diameter thereof and which is fixed to the pumping camshaft. The said pumping camshaft has single camlobes actuating the intake valves with the said diameter difference of the relevant gears providing for the increased cyclic velocity of the intake valves.

The method of operation including the valve timings of the alternative mode is now described with reference to a single

unit. The intake valves 12 begin to open when the pumping piston moves through to seventy pumping CA degrees after TDC. This allows the compressed intake air from the previous cycle to expand substantially to the pressure of the intake manifold before it opens. With the intake valves 12 opened and the pumping piston moving towards its BDC position, the intake air is induced into the pumping cylinder 5. Whilst the said piston 16 is moving towards BDC, the intake air within the crankcase is compressed. If the engine is operating above or about, fifty percent of its possible load, then the turbocharger 41 is operating efficiently, and as the pumping piston uncovers the crankcase transfer ports 34 at fifty said CA degrees before BDC, then no crankcase transfer occurs as the pressure in the said cylinder resulting from the turbocharger 41 is as high or higher than that of the said crankcase. This then provides for the said crankcase compression to be utilized at the lower loads but not at the higher loads as well as minimizing the maximum pressures attained in the said crankcase which then reduces the sealing requirment thereof and allowing for lighter said reed valve materials with lower opening pressures. As the said piston 16 moves through to fifty said CA degrees after BDC, the crankcase transfer ports 34 are closed and when the said piston moves to sixty said CA degrees after BDC, the intake valves 12 are closed and the induction of the intake air through the intake ports ceases whilst if the engine is operating at a low load then on the said pistons up stroke, intake air will be induced into the crankcase through the crankcase intake valves 32. One of the transfer valves opens when the pumping piston is at its BDC positon, to initiate the transfer phase to the respective power cylinder which the said transfer valve opens into. The said transfer valve then remains open untill the pumping piston has moved to ten said CA degrees after TDC and which is the same as that shown in FIG 6 and being forty five power CA degrees before the piston of the said respective power cylinder reaches TDC. The piston of the pumping cylinder then continues towards BDC and begins a new cycle thereof when the intake valves begin to open again at seventy pumping CA degrees after TDC whilst the intake air of the next said cycle is transferred to the other power cylinder

and so forth as is described hereinbefore.

During the first part of the transfer phase to the respective power cylinder, the secondary valve thereof is open providing for the scavenging of the exhaust gases from the combustion chamber thereof. The said secondary valve closes when the piston of that respective power cylinder has moved to one hundred and fifteen power CA degrees before TDC. During this time the exhaust valve of the said respective power cylinder is open and closes when the piston thereof has moved to forty five power CA degrees before TDC, allowing for nearly all the exhaust gas to be scavenged from the said cylinder except for a small residual portion thereof remaining. This is retained to highly pressurise the remaining gas so that when the secondary valve reopens when the piston thereof is at five power CA degrees before TDC, the pressure in the power cylinder is not significantly lower than that of the combustion chamber thereof which would decrease the thermal efficiency attainable. When the said piston is positioned about forty power CA degrees before TDC, diesel type fuel is injected into the said combustion chamber which results in combustion occuring just after the said relevant transfer valve has closed and so that as the said secondary valve thereof is opened, about fifty percent or more of the combustible mass has been combusted. With combustion completed and the said power piston moving towards BDC, the gas from the combustion chamber flows through the secondary port and open valve thereof to expand substantially to atmosperic before the exhaust valve of the said cylinder is opened when the piston thereof is at forty power CA degrees before BDC. This initiates the exhaust phase of the said cylinder and as the piston thereof moves towards TDC, it positively scavenges the said cylinder untill the next transfer phase thereinto begins which starts the next cycle thereof and as is described above. The operation of the other power cylinder has the same said valve and cyclic operation as that described above for the said respective power cylimder but as is obvious, it occurs 180 power CA degrees before and after it occurs in the said respective cylinder.

The alternative V configuration of FIG 7 has two units being in

the said configuration with each said unit being one bank of cylinders for the said engine whilst the pumping cylinders thereof are located to the inside of the said V, and of the power cylinders. A single pumping crankshaft 2 causes reciprocation of both said pumping pistons 16 whilst a single power crankshaft 1 causes reciprocation of all said power cylinders.

Obviously, many modifications and variations of the present invention are possible and it is therefore understood that within the scope of the appended claims, the invention may be practised otherwise than as specifically described.

CLAIMS:

1. A internal combustion engine comprising of one or more units with each unit consisting of; a pumping cylinder and two power cylinders all operating on two stroke cycles and all having reciprocating pistons operable therein; said pumping cylinder having a mainshaft causing reciprocation of the pumping piston therein and being driven to perform two cycles to a power cylinder single cycle; said power cylinders having a mainshaft to cause recriprication of the power piston in each said power cylinder; said power pistons being phased or being phased about one stoke apart; a cylinder head which closes adjacent ends of all said cylinders at the opposite end thereof to where the mainshafts are located; a said cylinder head which may extend down the walls of the said cylinders to where one third of the respective cylinder volome is provided; a combustion chamber volume for each power cyclinder is provided in the said head; said combustion chambers have atleast the major portion of combustion occuring therein and may be in constant communication with their respective power cylinders or communication therebetween may be timed by a secondary valve means whilst in either case, the said chamber is still refered to as the respective power cylinder, unless is otherwise stated; the said head has two transfer ports therethrough which communicate each power cylinder with the pumping cylinder at the head sections thereof and are timed atleast by transfer valve means which time communication between the said transfer ports and their respective power cylinder; the said head has two or more exhaust ports therethrough, with one or more of the said exhaust ports exiting each power cylinder whilst communication therebetween is timed by exhaust valve means to allow exhaust gas to flow from the said power cylinders as required by the engine operation; the said head has one or more intake ports therethrough and to the pumping cylinder, with intake valve means timing communication therebetween and to the pumping cylinder; said intake valves open to allow atleast a major portion of the intake charge to be induced into the pumping cylinder, substantially atleast when the piston thereof is moving to increase the said cylinder volume; the said intake charge may be any combination of air fuel and exhaust gas used

in the combustion process but has atleast sixty percent or more of the intake air used in said combustion; the intake charge having been induced into the pumping cylinder, is transferred alternately to each of the power cylinders through the respective transfer ports thereof and which have the transfer valves thereof open, and into the said power cylinders as the pumping piston, substantially atleast moves to decrease the volume thereof; after combustion the power pistons move through to BDC with the respective exhaust valve of a power cylinder opening as the piston thereof has moved to about, or before said position which initiates the exhaust phase thereof.

2. The engine of claim 1 wherein; the pumping piston leads the piston of the power cylinder which the intake charge is about to be transfered into, to the TDC position by less than 60 % of the time the said power piston is moving towards said position; the said means which cause the pumping piston to operate at the increased cyclic velocity, is driven from the mainshaft of the power cylinders or the engine output shaft.

3. The engine of claim 2 wherein; the pumping cylinder is positioned atleast substantially at an equal distance to each of the power cylinders; said intake valves begin to open between 0 - 20 % of the time required for a power cylinder cycle, after the pumping piston reaches its TDC position and closes between 20 - 48 % of the said time after the said position; the transfer valve, which times a respective transfer port to allow the transfer of the intake charge of that particular pumping cylinder cycle into a power cylinder, begins to open between 10 - 40 % of the said time before the said position and closes between 10 % of the said time before and after the said position; said exhaust valve of a power cylinder begins to open between 25 - 55 % of the said time after the piston thereof reaches its TDC position and closes between 55 - 97 % of the said time after the said respective power piston reaches TDC; the transfer valve of a power cylinder closes atleast before 30 % of the combustible mass thereof is combusted.

4. The engine of claim 3 wherein; a constant volume combustion chamber is provided between the said transfer valves and secondary valves; said combustion chamber providing

constant volume combustion for atleast a major portion of the time required for combustion and of the combustible charge; said secondary valves time communication between the combustion chambers and the power cylinders; the respective secondary valve of a said power cylinder begins to open between 10 % of the time required for a power cylinder cycle before and 40 % of the said time after the pumping piston attains its TDC position and closes between 40 - 0 % of the said time before the said postion.

5. The engine of claim 3 wherein; the combustion chambers remain in constant communication with their respective power cylinder.

6. The engine of claim 4 wherein; at least the transfer and exhaust valves are of the poppet type; the stems of the transfer valves and stem or other actuating portion of the secondary valves do not extend through the combustion chamber so that combustible mixture therein is not located all around the said portion of the said valves.

7. The engine of claim 5 wherein; at least the transfer and exhaust valves are of the poppet type; the head of the transfer valves is located atleast substantially axially above the cylinder but to one side thereof; the combustion chamber is located below the head of the transfer valves with the walls of said chamber extending substantially towards the mainshaft thereof so that the walls defining the said chamber act as a port and converge the flow of the transfering intake charge after the said transfer valves and direct it to leave the combustion chamber, in a downward direction; 70 % or more of the volume attainable when the power pistion is at TDC is available in the said combustion chamber.

8. The engine of claim 3 wherein; the axis of the main shaft of the pumping cylinder is located above the power cylinder main shaft axis on that line which extends therefrom and which is perpendicular to the power cylinder axis so that all said cylinders of a unit have a parallel axis and are in line.

9. The engine of claim 3 wherein; the pumping cylinder mainshaft has any said valve driving or actuating means and or other auxiliary device being driven from means provided or located thereon.

10. The engine of claim 3 wherein; the portion of the power cylinder mainshaft between the 2 power cylinders has a valve driving or actuating means and or other auxiliary device being driven by means provided or located thereon.

11. The engine of claim 6 wherein; all said cylinders of a unit, have a parallel axis; the secondary valves are of the poppet type; atleast the transfer and exhaust valves utilize the pressure respectively within the combustion chamber and power cylinder to increase the pressure on the sealing faces threrof; the transfer valve of a power cylinder has its stem extending from the head of the said valve so that it extends within, 15 degrees past and in the direction of the power cylinder fron the pumping cylinder, that plane which is perpendicular to a line drawn between and which is at right angles to the pumping and said power cylinder axis and, 15 degrees past and in the direction of the mainshaft, that plane which is perpendicular to the cylinder axis; the respective secondary valve of a power cylinder has its stem extending from the head of the said valve so that it extends within, 15 degrees past and in the direction of the pumping cylinder from the power cylinder, that plane which is perpendicular to the above said line which extends between the said pumping and power cylinders and, 15 degrees past and in the direction of the mainshaft, that plane which is perpendicular to the cylinder axis.

12. The engine of claim 7 wherein; all said cylinders have a parallel axis; The transfer and exhaust valves utilize the pressure within the combustion chamber power cylinder volume to increase the pressure on the sealing faces thereof; The respective transfer valve of a particular power cylinder has its stem extending from the head of the said valve so that it extends within, 15 degrees past and in the direction of the power cylinder from the pumping cylinder, that plane which is perpendicular to a line drawn between and which is at right angle to, the pumping and said power cylinder axis and, 15 degrees past and in the direction of the mainshaft, that plane which is perpendicular to the cylinder axis.

13. The engine of claim 3, substantially as herein described with reference to the drawing numbers 1 - 3, but except for;

the overhead cam gear and drive thereof; the specified fuel and ignition means; the specified crankshaft and conrod design; the component locations but foregoing that of the combustion chamber, the valves, and of the ports thereof.

14. The engine of claim 3, substantially as herein described with reference to the drawing numbers 4 - 6, but except for; the overhead cam gear and drive thereof; the specified fuel and ignition means; the specified crankshaft and conrod designs; the component locations but foregoing that of the combustion chamber, the valves, and of the ports thereof.

15. The engine of claim 3 wherein; 2 or more said units are utilized in a V configuration so that the atleast the power cylinders thereof are in a V configuration with atleast the power cylinders of one or more units being a bank of cylinders in said V configuration; a singular mainshaft causes reciprocation of the pistons of the power cylinders of atleast the 2 units required for said V configuration and a seperate singular mainshaft causes reciprocation of the pistons of the pumping cylinders of atleast the said 2 required.

16. The engine of claim 3 wherein; atleast the pumping cylinder utilizes the crankcase thereof for crankcase compression with crankcase transfer ports communicating the said crankcase with the lower said cylinder wall so that as the piston thereof is near its BDC position, it uncovers the said ports to allow the flow of a portion of the intake charge into the said cylinder; crankcase intake valve means time communication between the crankcase intake port or manifold and the said crankcase so that the said charge is induced whilst the piston thereof is atleast substantially moving towards its TDC position.

17. The engine of claim 3 wherein: the pumping cylinder is positioned other than between the power cylinders so that the distance between the said power cylinders is less than the sum of one pumping cylinder bore diameter and 2 wall thickness which may seperate the pumping and power cylinders; the exhaust manifold of each power cylinder enters a turbocharging device which utilizes the exhaust gases of the said power cylinders to increase the pressure of the intake gas.

18. The engine of claims 16 and 17 combined and wherein; the

pressurized intake manifold leading from the turbocharger communicates with the pumping cylinders intake port; the said crankcase intake port and or manifold is naturally aspirated.

19. The engine of claim 3 wherein; the intake valves closes 25 - 40 % of the said time after the said position; the said transfer valve opens 15 - 40 % of the said time after the said position.

20. The engine of claim 3 wherein; poppet type valves are used for atleast the transfer and exhaust valves; a variable valve timing mechanism which may vary any combination of the said valves open and or closing times and rates as well as open duration and or lift, is used to vary atleast the closing time of the exhaust valves.

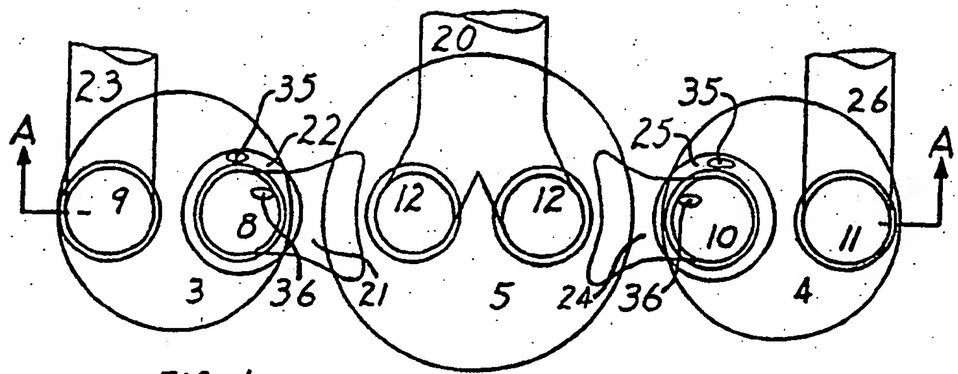

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FIG-1.

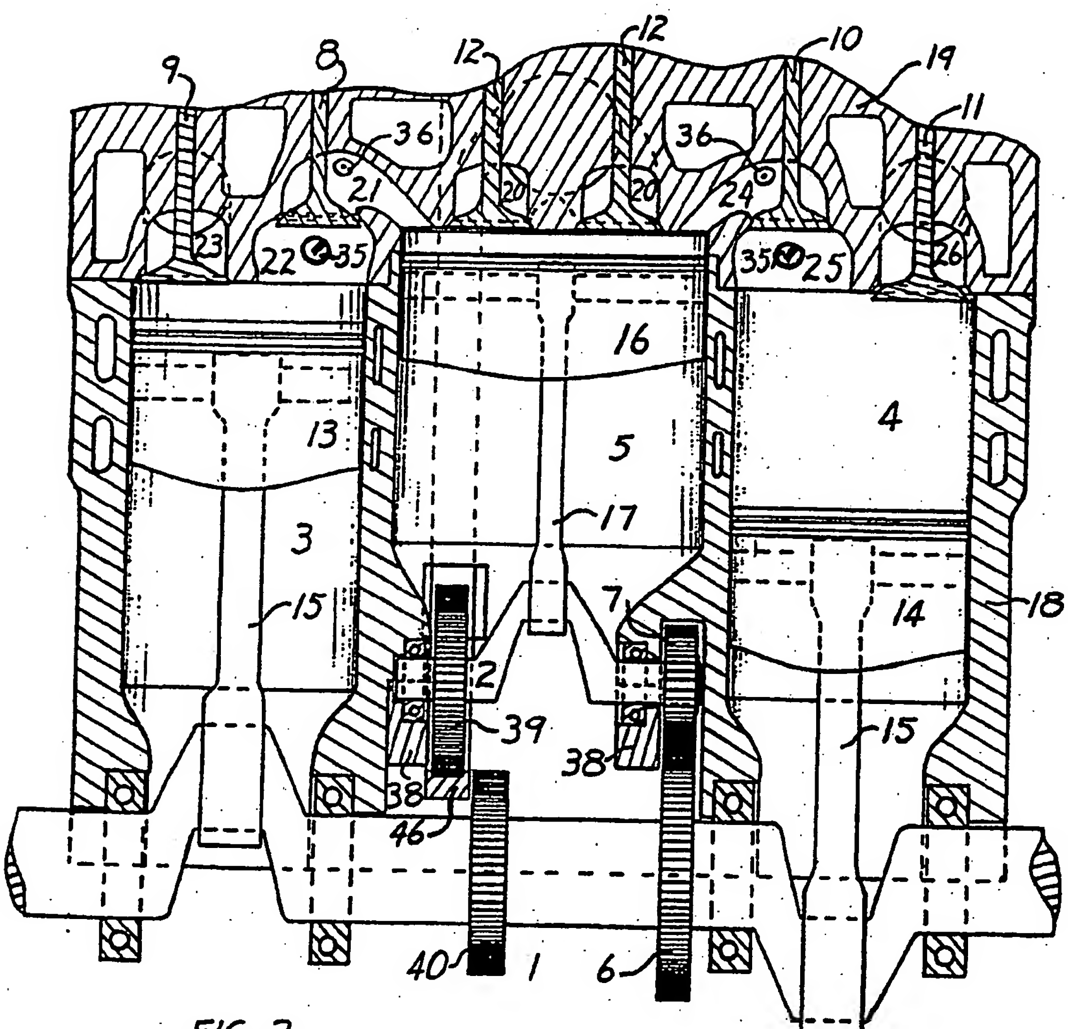

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FIG-2.

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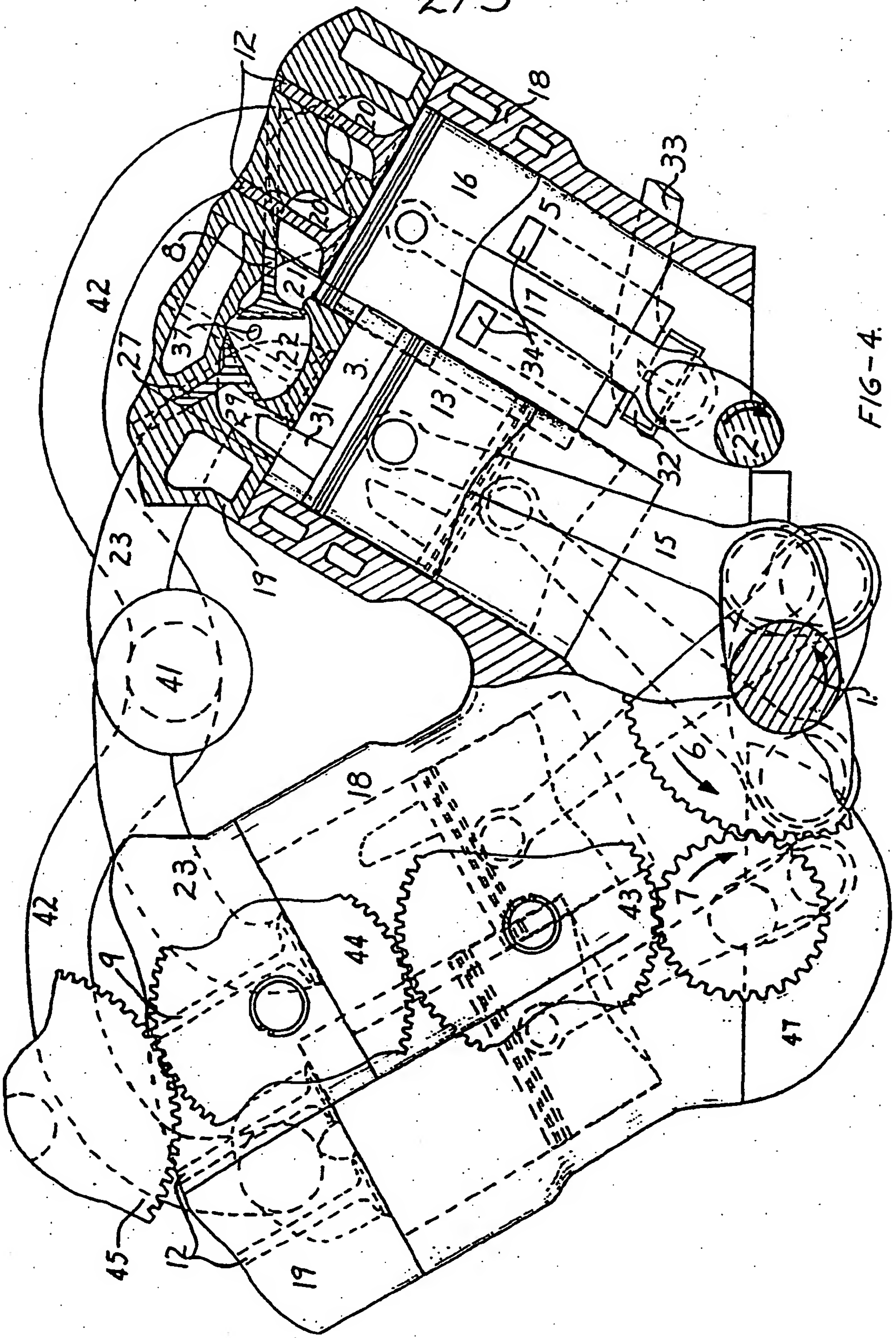
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FIG-4.

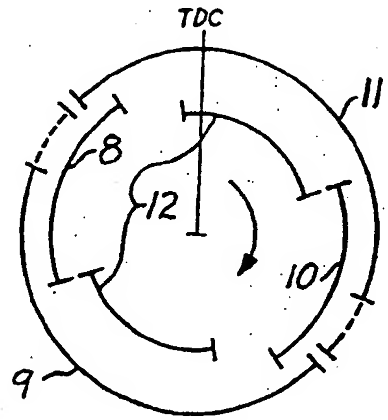

FIG-3

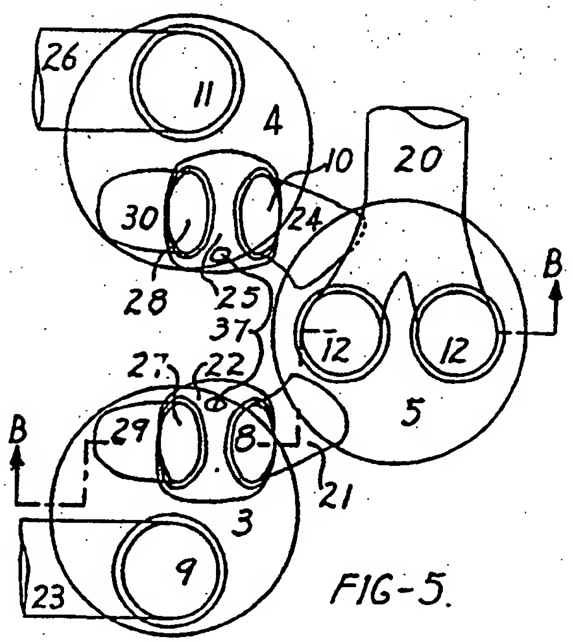

FIG-5.

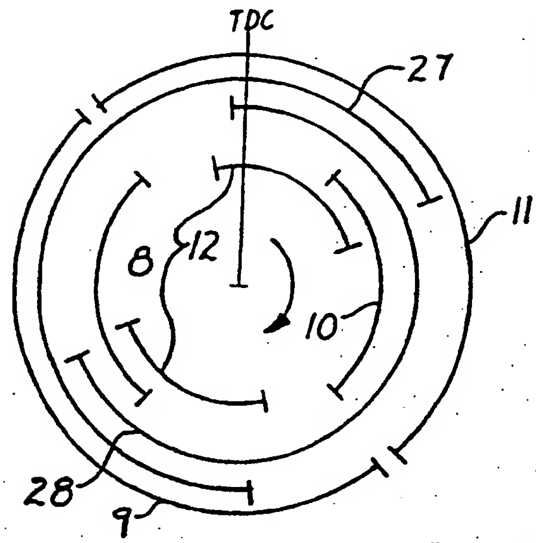

FIG-6.

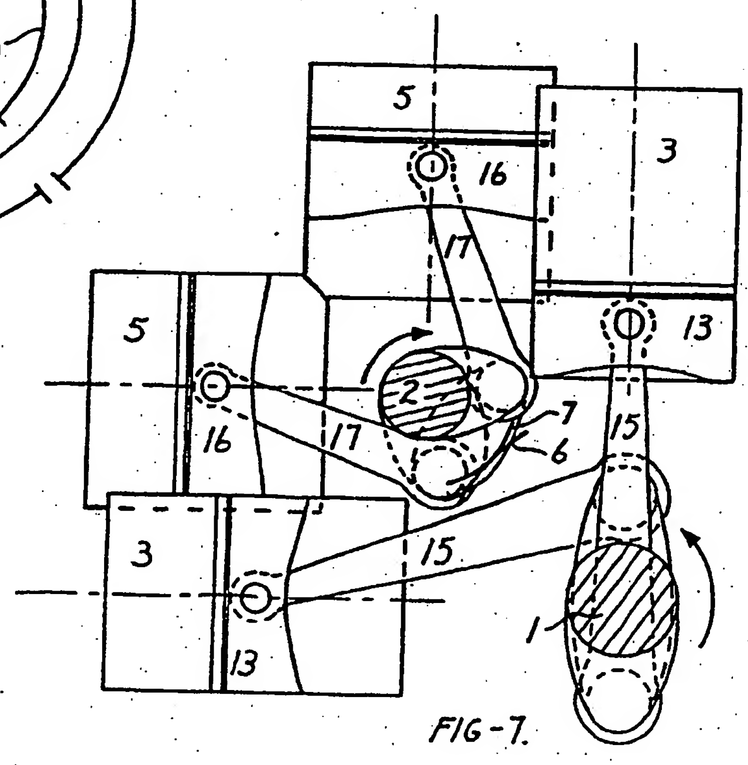

FIG-7.

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) 6

According to International Patent Classification (IPC) or to both National Classification and IPC

Int. Cl.[5] F02B 33/06, 33/20, 33/22

II. FIELDS SEARCHED

Minimum Documentation Searched 7

Classification System	Classification Symbols
IPC	F02B 33/06, 33/20, 33/22

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched 8

AU: IPC as above, Australian Classification 64.2

III. DOCUMENTS CONSIDERED TO BE RELEVANT 9

Category*	Citation of Document, 11 with indication, where appropriate, of the relevant passages 12	Relevant to Claim No 13
A	US,A, 1881582 (HOLLOWAY) 11 October 1932 (11.10.32)	
A	Patents Abstracts of Japan, M644, page 146, JP,A, 62-135615 (FUJI HEAVY IND LTD) 18 June 1987 (18.06.87)	
A	GB,A, 2071210 (KALTENEGGER) 16 September 1981 (16.09.81)	
A	AU,B, 21473/48 (143571) (RICARDO) 22 July 1948 (22.07.48)	
A	AU,B, 18746/56 (213330) (HAMMICK) 5 December 1957 (05.12.57)	
A	GB,A, 169799 (GERNANDT) 3 October 1921 (03.10.21)	

* Special categories of cited documents: 10

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
8 August 1990 (08.08.90)	21 August 1990
International Searching Authority	**Signature of Authorized Officer**
Australian Patent Office	C M WYATT

Form PCT/ISA/210 (second sheet) (January 1985)